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DESIGN RECOMMENDATIONS FOR R-718 HEAT PUMPS IN HIGH TEMPERATURE APPLICATIONS

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ABSTRACT

The transition towards carbon neutral industry requires sustainable and climate friendly heat supply. Heat pumps can meet these requirements, but currently achievable temperatures for supply of process heat are limited to around 100 °C. Using water (R-718) as refrigerant shows a good thermodynamic and environmental performance at higher temperatures, while a sophisticated design is required to compensate increased investment cost due to relatively large volume flows.

This work analyses the design of two-stage R-718 heat pumps with turbo compressors with a focus on desuperheating and suggests design recommendations to ensure a competitive thermodynamic and economic performance. The results have shown that the economic performance becomes especially competitive at evaporation temperatures above 100 °C with COPs of up to 4.3 and specific investment cost below 250 €/kW of supplied heat. The suggested approach to realize the desuperheating was found as the optimal solution with respect to pressure drop and space requirements.

Keywords: Natural refrigerant, Water heat pump, Turbo compressor, Desuperheating, Intercooling, Industrial heat supply

1. INTRODUCTION

The European Union moves towards carbon emission neutrality by 2050 and many countries have consequently committed to transform their industry sectors becoming independent of fossil fuels. Wolf and Blesl (2016) have shown that current state of the art heat pump technology could reduce the final energy consumption of the industrial sectors of EU-28 by 15 % and the energy-related CO₂ emissions by 17 %, considering the current average CO₂ emissions related to the electricity production. This increases the demand and attractiveness of heat pumps in an industrial context.

Several studies have analyzed the possibilities to integrate heat pumps for supply of process cooling and heating in industrial processes and were documented by the work of Annex 35 and 48 of the IEA Heat Pump Centre (2014). The realized projects demonstrated the possibility of a good economic and thermodynamic performance. While it can be noted that heat pump equipment is readily available and a proven technology for supply temperatures below 100 °C, different studies (e.g. Dupont and Sabora (2009); Wolf et al. (2014)) identified considerable process heat requirements with supply temperatures above 100 °C.

Different reports (e.g. Elmegaard et al. (2017); IEA Heat Pump Centre (2014)) documented several projects, in which the technical feasibility of heat pumps with supply temperatures above 100 °C was demonstrated. It can be assumed that a widespread application of high temperature heat pumps is limited by generally lower maximum obtainable performances and challenging economic boundary conditions. This induces the research

and development activities to focus on solutions with a high thermodynamic performance at minimum investment cost.

The performance of a heat pump – as well as the design of the components and thus the investment cost – are strongly dependent on the choice of the working fluid. Chamoun et al. (2011) and Larminat and Arnou (2012) compared different synthetic and natural refrigerants in different cycle layouts for a range of high temperature applications. They found water (R-718) to show the most promising thermodynamic performance, besides having other favorable refrigerant characteristics, such as being environmentally friendly, non-flammable, non-toxic and available at low cost.

Irrespective of the promising thermodynamic performance, the utilization of water as refrigerant implies peculiarities, which impose challenges to an economically effective design. Compared to typically used refrigerants, the critical pressure and temperature of water are relatively high. This results in low pressures and high volume flow rates at the considered temperature ranges. Especially the comparably low volumetric heating capacity of R-718 requires the compression equipment to have volumetric capacities, which are multiple times higher than required for other refrigerants. Since the compressor is typically a major contribution to the overall investment cost of the heat pump installation, this has a strong impact on the economic performance. On the other hand, water can be used both as refrigerant and heat transfer medium, which implies possibilities for direct heat exchange with small temperature differences and pressure drops.

In order to facilitate the market introduction of R-718 heat pumps, different research projects have focused on the development of cost effective compressors. The PACO project, Chamoun et al. (2014); Larminat et al. (2014), followed the findings from Chamoun et al. (2011) and Larminat and Arnou (2012) to meet the application potentials shown by Dupont and Sapora (2009). The project aimed to develop a compression technology to supply 700 kW thermal load at a temperature lift between an evaporation temperature of 90 °C and a condensation temperature of 130 °C using R-718. Chamoun et al. (2014) presented numerical and experimental studies of a liquid injection screw compressor technology, while Larminat et al. (2014) presented a modified two-stage centrifugal compressor. The compressor was designed with magnetic bearings and two impellers at the same motor shaft. The experiments have shown COPs above 5 at a temperature lift of 40 K at a frequency of 665 Hz. The authors expect to obtain a COP around 5.5 with a more sophisticated development.

Madsboell et al. (2015) presented the development of a centrifugal turbo compressor for industrial applications with a temperature lift of 25 K at evaporation temperature in the range of 90 °C to 110 °C and a capacity of 100 kW to 500 kW heat supply. The developed compressor was based on automotive superchargers and designed for rotational speeds of up to 100,000 rpm with a power consumption of 50 kW.

Bantle (2017a) described the modular construction of the compressor, the gear, the motor and the inverter, which allows adjusting the unit to different applications by utilizing different impellers, as developed by Madsbøll. Pilot test results were presented with satisfying performances. The modular construction and the possibility of exchanging the impeller according to the application cover a broad range of applications with different supply temperatures, temperature lifts and capacities. Bantle (2017b) presented the design of a test rig of a two-stage vapor compression cycle using the described compression technology. He furthermore outlined the requirement of efficient desuperheating at intermediate pressure in multi-stage systems, to keep the temperatures in the high pressure compression stages at a moderate level.

This study investigates the design of a R-718 heat pump using turbo-compressors with the focus on effective desuperheating, which is preferably realized as a cost-effective and compact unit with a minimum pressure drop. The work comprises a description of the available compression technology and the design requirements for multi-stage compression systems, before different desuperheating technologies are introduced and evaluated.

2. METHODS

2.1 Description of available compressor technology

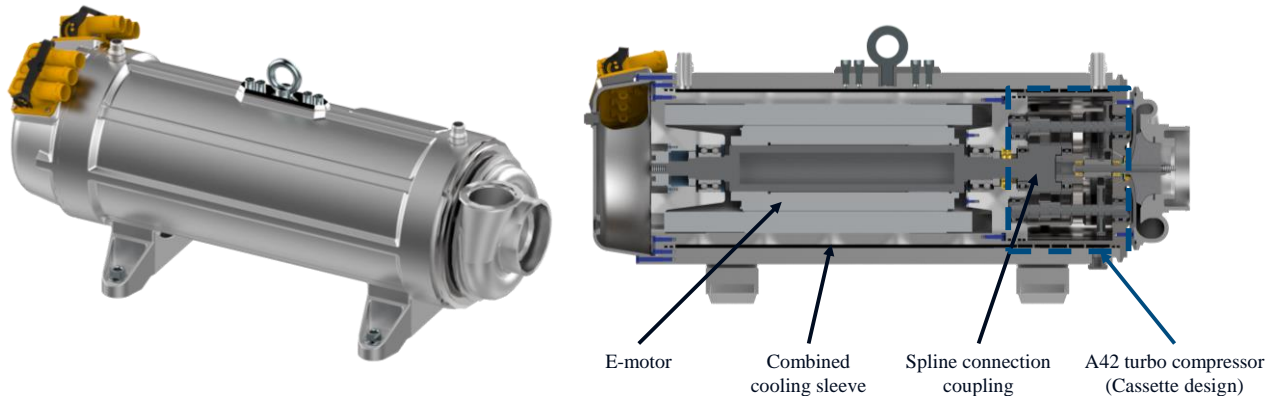


Figure 1: 3D-model (left) and sectional view (right) of the unit evaluated in the current study including electric motor, gear and compressor as presented by Bantle (2017b)

This sub-section gives an overview of the available compressor technology, which constitutes the core of the modular concept. Figure 1 shows the compressor unit considered in this study. The unit consists of an electrical DC-motor, a lifetime lubricated gear and an exchangeable turbo-compressor cassette. The motor is powered by an inverter enabling continuous frequency control of the compressor. The compressor can be equipped with an impeller size specifically designed for the application.

Currently there are two impeller types, which were designed for operation in a two-stage arrangement for different volume flow rates. The design conditions and performances of the two impeller types are shown in Table 1. The inlet conditions assume the steam to be 10 K superheated. The isentropic efficiencies of the compressor are expected values, which were validated in air tests and for comparable equipment operating with steam, Bantle (2017a). There are additional efficiencies of 98 % for the inverter, 95 % for the motor and 95 % for the gear, to account for mechanical inefficiencies causing heat losses to the environment and the cooling system.

Table 1: Operation parameters at design conditions of two impeller types as presented by Bantle (2017b)

	Impeller 1: Low pressure stage	Impeller 2: High pressure stage
Inlet conditions:	$\dot{m}_{in} = 0.196 \text{ kg/s}$ $\dot{V}_{in} = 1231 \text{ m}^3/\text{h}$ $p_{in} = 1.0 \text{ bar}$ $T_{in} = 110 \text{ }^\circ\text{C}$	$\dot{m}_{in} = 0.222 \text{ kg/s}$ $\dot{V}_{in} = 468 \text{ m}^3/\text{h}$ $p_{in} = 3.2 \text{ bar}$ $T_{in} = 145 \text{ }^\circ\text{C}$
Rotational speed:	$n = 80,000 \text{ rpm}$	$n = 80,000 \text{ rpm}$
Pressure ratio:	$PR = 3.2$	$PR = 2$
Expected isentropic efficiency:	$\eta_{is} = 0.74$	$\eta_{is} = 0.74$

2.2 Two-stage vapor compression cycles

The presented compression technology can be applied in closed loop heat pumps or open heat pumps, so called mechanical vapor recompression (MVR) systems. In such systems, the vapor is directly taken from a process, compressed and used to evaporate or re-heat liquid from the same process. The MVR technology shows an increased performance due to reduced temperature differences during heat transfer but can be unfeasible in some applications. Since the closed loop heat pumps are independent of the process a higher market share than for MVR applications is expected. This study thus focused on the design of a closed loop heat pump.

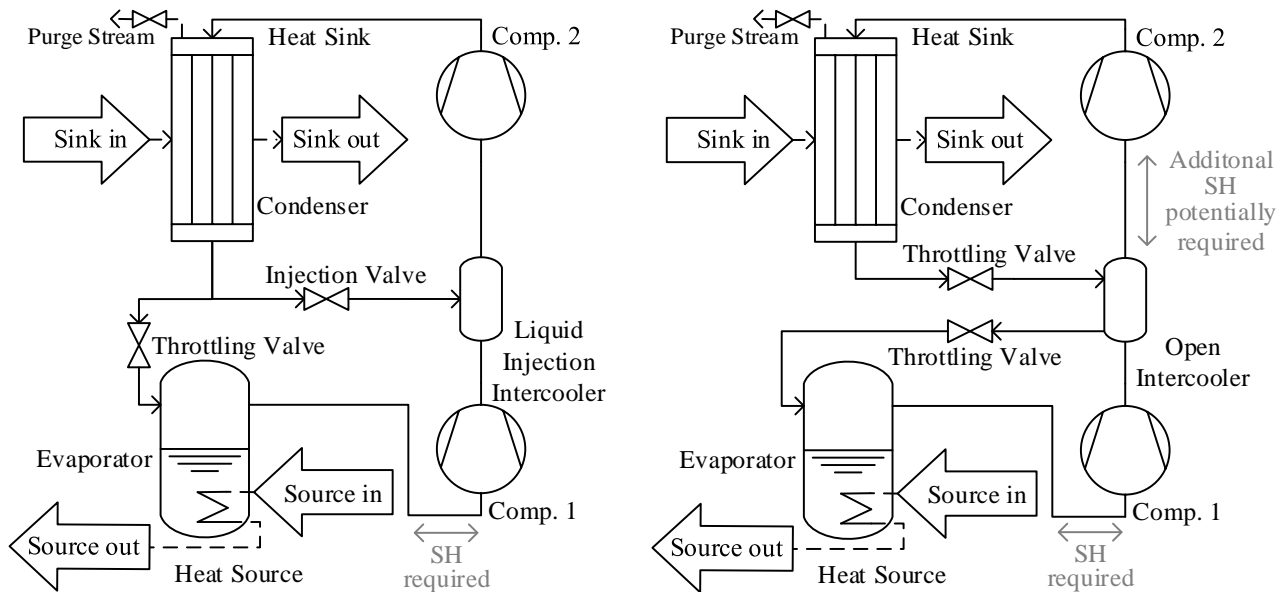


Figure 2: Layout of two-stage R-718 vapor compression heat pump with: Parallel expansion and liquid injection intercooling (left) and serial expansion and an open intercooler (right), SH = Superheating

Figure 2 shows two possible layouts of a two-stage closed loop heat pump. In both cycles, the working fluid evaporates while receiving heat from the heat source, before being compressed in two stages and condensed at high pressure while transferring heat to the heat sink. The cycle with parallel expansion (left) cools the superheated vapor after the first compression stage by injecting a defined amount of saturated liquid from the condenser, while the cycle with serial expansion (right) uses the complete stream from the condenser.

Since the turbo compressors are sensitive to damages due to liquid droplets, systems are typically designed with a minimum superheating of 5 K to 10 K. The required superheating can be realized by i) a suction line heat exchanger, which uses heat from subcooling the liquid after the condenser, ii) recirculating part of the compressed gas from the outlet and mixing it into the suction line, or iii) external heating, such as heat tracing.

While one of these solutions is required in any case in front of the first stage, it might not be required for the second stage. The liquid injection cycle, as well as some of the open intercooler solutions, are able to control the outlet to a defined temperature, and thereby omit the requirement of a superheating device.

Conventional open intercoolers do however typically supply steam, which is saturated or closed to saturated conditions, and therefore require the implementation of one of the above mentioned measures to ensure sufficient superheating at the inlet of compressor 2. While the liquid injection obviates the requirement of these additional measures, the open intercooler cycle does in turn economize the complete amount of flash gas generated during the first throttling stage and has therefore benefits with respect to the cycle performance. The choice of the cycle layout depends accordingly on the chosen equipment to realize the desuperheating.

In order to compare different desuperheating technologies, a model for the thermodynamic cycles was implemented in Engineering Equation Solver (EES), Klein (2017). The model consisted of energy and mass balances and considered the above mentioned efficiencies for the compressor and the remaining components. The choice of the superheating technology is dependent on further aspects, such as behavior during transient operation and economic aspects, and therefore excluded from the presented numerical analysis.

The analysis was conducted for a range of evaporation temperatures between 80 °C and 110 °C. The lower compression stage was assumed to operate at 80,000 rpm, while the second compression stage was defined to operate at a rotational speed to yield the volume flow rate given by the first compression stage. It was furthermore assumed that both compression stages achieved the pressure ratio and isentropic efficiency as in the design case. The volume flow rate at the inlet of compressor 1 was assumed to be the same as at design conditions. The maximum rotational speed of the unit is 80,000 rpm.

The total investment cost of a closed loop two-stage heat pump system using the described compressor technology and a system designed to deliver 700 kW at 110 °C evaporation temperature is expected to be around 125,000 €, of which 50,000 € are required for the two compressors including motor, gear and inverter. The estimation was based on experiences from constructing a test rig and a preliminary assessment of the components shown in Figure 2.

2.3 Possibilities to realize the desuperheating

A good overview of the basics and current technologies for steam cooling as used in steam supply networks is given by Spirax Siraco (2012). An overview of the commonly used technologies for intercooling in multi-stage compression refrigeration and heat pump systems is given by Stoecker (Ch. 3.5-3.6, pp. 72-77, 1998).

There are basically two approaches to realize the desuperheating. Either liquid can be injected into the gas stream, which is cooling it down while evaporating, or the gas stream can be bubbled into a liquid holdup, which creates a good heat transfer between the gas and the liquid, resulting in vapor being saturated at the outlet. Figure 3 shows sketches for the desuperheating technologies considered in the analysis. Solutions b), c) and d) can be used in the open intercooler cycle and therefore have a drainage line, which is not considered when used as an intercooler.

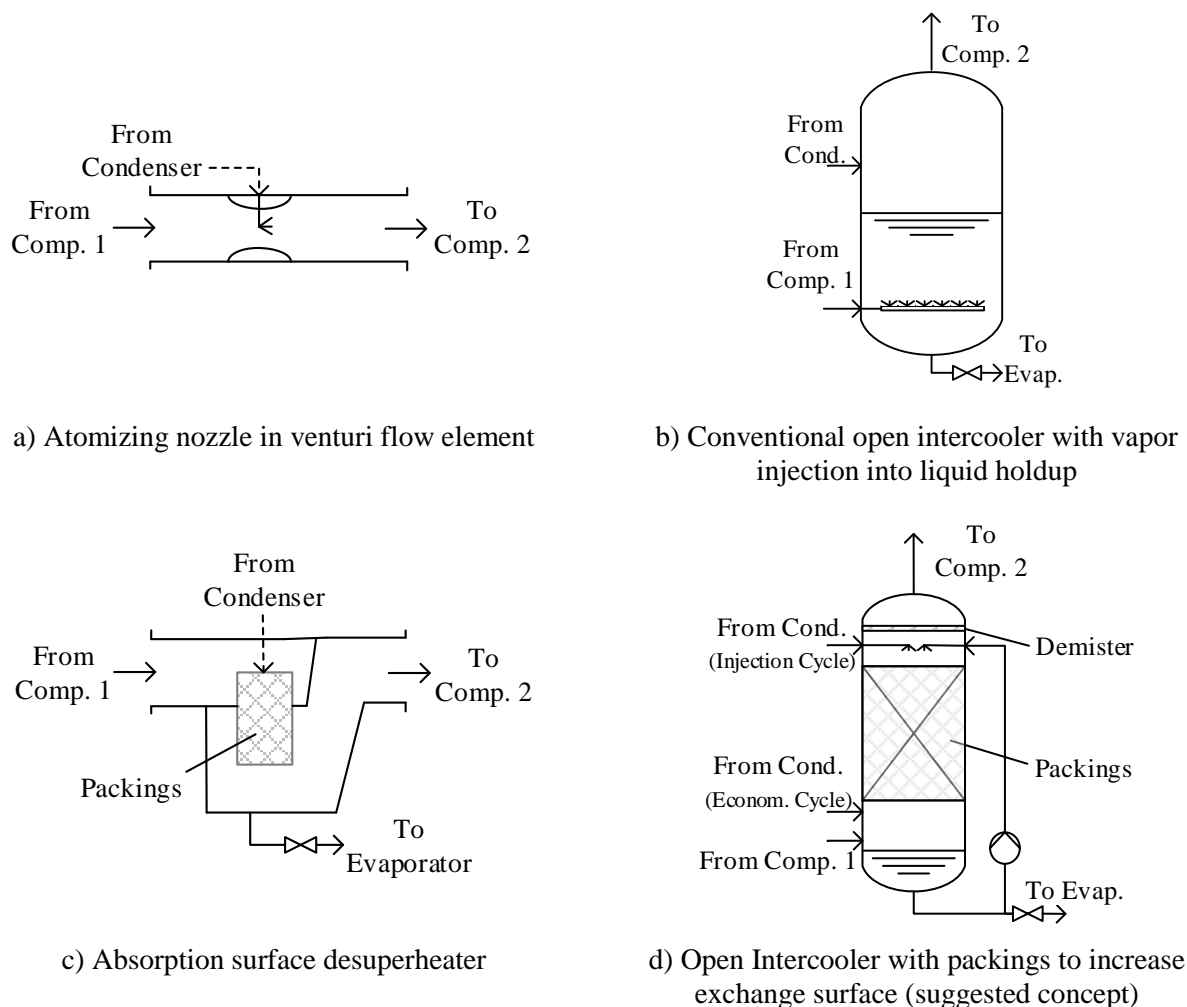


Figure 3: Sketch of different technologies for desuperheating

The steam at the outlet of the intercooler should desirably be superheated by approximately 5 K to 10 K to ensure safe operation and to keep the outlet temperature of compressor 2 as low as possible. On the other hand, the formation of condensate droplets must be prevented to protect the turbo-compressor, which could be induced by heat losses in the suction line piping. Therefore, appropriate measures to avoid droplet formation,

have to be considered, in case the desuperheating technology yields saturated steam and is not capable of controlling the temperature.

a) Liquid injection atomizing nozzle: The liquid injection is a technology which is commonly used in steam supply networks of industrial plants, in which the steam is distributed in superheated conditions, before it is cooled down to saturation temperature close to the consumer.

The length of the pipes has to be long enough to give sufficient time for the liquid droplets to evaporate in the gas phase. The length is influenced by several parameters such as temperature and pressure of the gas and the injected liquid flow, as well as by the flow conditions. Spirax Siraco (2012) outlines the benefits of turbulences in the flow and recommends velocities of up to 60 m/s. The resulting pipe lengths are accordingly long and can be obtained from guidelines, such as Spirax Siraco (2012) or Schutte & Koerting (2017). The nozzles can be arranged in different configurations such as on the periphery or directly in the stream. The nozzle can be oriented co- or countercurrent to the stream and the stream might be accelerated by a decreased cross section area. The pressure drop calculations assume an instantaneous evaporation of the injected liquid and are calculated according to VDI Heat Atlas (Ch. L1.1, pp. 1055-1056, 2010).

b) Open intercooler: The open intercooler is commonly used in heat pump and refrigeration systems and the benefits are an efficient desuperheating, the complete recovery of the gas formed during expansion and a compact construction. Stoecker (Ch. 3.5-3.6, pp. 72-77, 1998) recommends the liquid holdup above the vapor injection to be 0.6 m to 1.2 m, which corresponds to a hydrostatic pressure of 0.06 bar to 0.12 bar for R-718.

c) Absorption surface desuperheater: The absorption surface desuperheater is developed and presented by Schutte & Koerting (2017). It injects the liquid into the vapor, which is conducted through a duct of wetted reaction rings. It is recommended for applications with limited availability of space. The unit requires a minimum of maintenance but introduces a higher pressure drop than the liquid atomizer solutions presented in point a). The unit enables furthermore the total recovery of the flash gas formed during expansion.

d) Open intercooler with packings: This solution utilizes packings, which are typically used in packed columns for separation processes, in order to create a large heat and mass transfer area at a minimum pressure drop. The packings are available in many different shapes and materials. VDI Heat Atlas (Ch. M7, pp. 1327-1342, 2010) and Kraume (Ch. 13, pp. 419-447, 2012) give an overview of dimensioning such apparatuses. When the unit is used as an intercooler, the stream from the condenser should be injected above the packings, while it should be injected below when utilized in the open intercooler cycle. Since the liquid should in both cases flow downwards while the steam is flowing upwards, a recirculation pump is added for operation as open intercooler.

The so-called flooding point at which an increasing gas velocity would stop the liquid flooding down defines the minimum diameter of the unit. This minimum required diameter should be exceeded by a certain safety margin to increase the range of stable operation. The ratio of the gas velocity to the maximum possible velocity at the flooding point is described by the flooding factor. The pressure drop and the flooding point were determined as described in Engel (2000) and Stichlmair et al. (1989) considering stainless steel packings with a specific surface of $160 \text{ m}^2/\text{m}^3$ and a voidage of 98 %. The friction factor was calculated according to VDI Heat Atlas (Ch. L2.6, pp. 1169-1177, 2010). According to Kraume (Ch. 13, pp. 419-447, 2012), the heat transfer was calculated analogously to convective heat transfer between a gas and a spherical body. The active heat transfer area was assumed as the entire surface of the packings and the heat transfer coefficient of the gas phase to be the dominating factor.

3. RESULTS

3.1 Results from cycle simulations of 2-stage cycle

The described cycle model was used to evaluate the performance at different operating conditions. The evaporation temperature was varied between 80 °C and 110 °C, while the pressure ratios and the efficiencies in the compressors, as well as the suction volume flow rate at the inlet of compressor 1 were assumed to be constant as defined in the design point. The volume flow rate at compressor 2 was given according to the inlet conditions defined by the first compression stage. The amount of injected saturated liquid was determined to yield an intercooling to 10 K above the saturation temperature of the intermediate pressure.

Table 2 shows an overview of the obtained results. While the COP increases with decreasing evaporation temperatures, the capacity decreases. This means that the unit can supply less heat at a better COP. While the COP increased by 12 % between an evaporation temperature $T_{\text{evap}} = 110^\circ\text{C}$ to 80°C , the capacity decreased by 39 %. Assuming a fixed investment, the specific investment cost per unit of supplied heat decreases from 484 €/kW at $T_{\text{evap}} = 80^\circ\text{C}$ to 179 €/kW at $T_{\text{evap}} = 110^\circ\text{C}$. The performance of the open intercooler cycle was approximately 3 % higher.

The injected mass flow rate varied from 10 % of the mass flow rate from compressor 1 at $T_{\text{evap}} = 80^\circ\text{C}$ to 12 % at $T_{\text{evap}} = 110^\circ\text{C}$. The superheated vapor was in all cases approximately 120 K above the saturation temperature at the outlet of compressor 1 and 82 K at the outlet of compressor 2.

The pressure drops in the evaporator and the intercooler were varied to analyze their influence on the performance. It was found, that a pressure drop of 0.1 bar in the evaporator decreases the condensation temperature by 3 K, while the same pressure drop in the intercooler results in a decrease of the condensation temperature by 1 K. The supplied heat load was found to be relatively independent of the pressure drop.

An variation of the difference between the temperature after intercooling and the according saturation temperature, which is achieved by a variation of the injected mass flow rate, showed a minor influence on COP and supplied heat load, with a slight increase of COP for smaller remaining superheating.

Table 2: Results from simulation of the injection cycle

Evaporation temperature T_{evap} :	80 °C	90 °C	100 °C	110 °C
Evaporation pressure p_{evap} :	0.47 bar	0.70 bar	1.01 bar	1.43 bar
System Performance				
Condensation temperature T_{cond} :	133.9 °C	147.8 °C	161.9 °C	176.2 °C
Condensation pressure p_{cond} :	3.03 bar	4.49 bar	6.49 bar	9.17 bar
Supplied heat load \dot{Q}_{sink} :	258.2 kW	368.4 kW	513.2 kW	699.7 kW
Specific total invest. cost per supplied heat load:	484 €/kW	339 €/kW	244 €/kW	179 €/kW
Coefficient of performance COP:	4.61	4.44	4.27	4.12
Compressor 1: PR = 3.2, $\eta_{\text{is}} = 0.74$				
Volume flow rate at inlet $\dot{V}_{\text{comp1,in}}$:	1231 m ³ /h	1231 m ³ /h	1231 m ³ /h	1231 m ³ /h
Mass flow rate \dot{m}_{comp1} :	0.100 kg/s	0.145 kg/s	0.204 kg/s	0.283 kg/s
Power consumption of inverter $\dot{W}_{\text{inverter1}}$:	33.3 kW	41.2 kW	71.0 kW	100.3 kW
Outlet temperature $T_{\text{comp1,out}}$:	232 °C	245 °C	258 °C	271 °C
Compressor 2: PR = 2, $\eta_{\text{is}} = 0.74$				
Volume flow rate at inlet $\dot{V}_{\text{comp2,in}}$:	470 m ³ /h	472 m ³ /h	474 m ³ /h	476 m ³ /h
Mass flow rate \dot{m}_{comp2} :	0.111 kg/s	0.160 kg/s	0.227 kg/s	0.316 kg/s
Power consumption of inverter $\dot{W}_{\text{inverter2}}$:	22.8 kW	33.9 kW	49.1 kW	69.7 kW
Outlet temperature $T_{\text{comp2,out}}$:	216 °C	230 °C	244 °C	258 °C

3.2 Design of intercooling units

a) Liquid injection atomizing nozzle: The liquid injection was dimensioned according to Spirax Siraco (2012) and required a pipe of 13 m with an inner diameter of 50 mm. The pressure drop was estimated to be 36 mbar at design conditions. Preliminary measurements have shown that an additional significant pressure drop of 40 mbar to 60 mbar is induced by the venturi injection unit and the required liquid separator. The measured pressure drops are relatively low, compared to typical applications in steam supply networks, Spirax Siraco (2016), and can only be achieved with sufficiently large equipment.

b) Open intercooler: The open intercooler can be realized with compact dimensions, but induces a pressure drop of up to 120 mbar, according to the hydrostatic pressure of the recommended liquid holdup of 0.6 m to 1.2 m., Stoecker (Ch. 3.5-3.6, pp. 72-77, 1998). According to Spirax Siraco (2012), the open intercooler provides typically saturated steam.

c) Absorption surface desuperheater: The surface absorption desuperheater has similar compact dimensions as the open intercooler. Schutte & Koerting (2017) reported higher pressure drops, up to 50 %, compared to

venturi nozzles, which corresponds to values up to 90 mbar, considering the measured values of the currently tested venturi nozzle with sufficiently large dimensions, or even higher values.

d) Open intercooler with packings: The open intercooler with packings was evaluated by applying semi empirical correlations for pressure drop and the heat transfer coefficients. The diameter was determined in order to not exceed the maximum velocity defined by the flooding point. At evaporation temperatures of 80 °C to 110 °C, an inner diameter of 250 mm yields a flooding factor of 10 % to 20 %, respectively. The heat transfer coefficients of the vapor vary from $80 \text{ W} \cdot (\text{m}^2 \text{ K})^{-1}$ to $160 \text{ W} \cdot (\text{m}^2 \text{ K})^{-1}$ for the same evaporation temperatures, which results in required packing heights of 0.6 m to 0.8 m, respectively. The pressure drop of the gas flow increased from 2 mbar at $T_{\text{evap}} = 80 \text{ °C}$ to 8 mbar at $T_{\text{evap}} = 110 \text{ °C}$. Considering some space for additional safety margins and a demister, the unit is expected to be realizable in a pipe of less than 1.5 m to 2 m length, which could be realized in a vertical pipe and is therefore consistent with the approach of constructing a modular and compact unit.

Table 3: Summary of different technologies for desuperheating

	a) Liquid injection	b) Conventional open Intercooler	c) Absorption surface desuperh.	d) Open Intercooler with packings
Pressure drop in design point:	40 mbar - 60 mbar (venturi injection) + 36 mbar (piping)	60 mbar to 120 mbar	Up to 90 mbar (or higher)	5 mbar
Dimensions for presented case:	Horizontal pipe: 13 m x ø 50 mm	Compact dimensions (h < 1.5 m)	Compact dimensions	Vertical pipe: 2 m x ø 250 mm
Complexity of construction:	Established technology	Established technology, simple control	Commercial product	Suggested concept with proven components
Recovery of flash gas:	Not possible	Complete recovery possible	Either complete recovery of flash gas, or control of outlet temperature possible	Complete recovery possible
Control of outlet temperature:	Possible	Not possible		Possible

Table 3 summarizes relevant aspects of the different available technologies. All units, except the open intercooler, can be used to regulate the temperature of the vapor at the outlet of the unit when utilized in the intercooler cycle, which implies that a defined mass flow rate of liquid is injected into the stream. Considering the application in an open intercooler cycle, the liquid injection nozzles are eliminated as a possibility, since the construction cannot handle too much liquid. The absorption surface desuperheater will furthermore lose the possibility to control the outlet temperature, and would require additional measures, such as external heat tracing of the suction lines, to ensure sufficient superheat at the inlet of the second compression stage. This means that the suggested concept with packings is the only solution, which can satisfy both requirements, as it is suitable for operation as an open intercooler while being able to control the vapor outlet temperature. This case requires an additional recirculation pump to feed a defined liquid stream to the top.

4. DISCUSSION

The cycle calculation assumed the efficiencies to be the same as in design conditions. While the first compressor stage worked with the same volume flow rate and pressure ratio as in design conditions, the second stage was adjusted to the volume flow rate as determined by the system and therefore operated at conditions deviating from the design conditions. It is therefore expected that the assumption of the compressors operating with the same isentropic efficiency as in the design point, is more applicable for the first compression stage, while increased uncertainties are expected for the second stage. In order to eliminate these uncertainties, a description of the performance of the impellers over the complete range of operating conditions working with

steam has to be determined and integrated into the simulation. This can be realized by a validated and calibrated model of the impeller or by completely measured compressor performance maps.

The results have shown the tendency of a decreasing performance and an increasing capacity for higher evaporation temperatures, while the increasing capacity had a dominating effect on the economic performance. Due to the fixed investment cost, the specific investment cost behaves similarly showing values as low as 179 €/kW. According to Wolf et al. (2014) a specific investment cost of up to 250 €/kW supplied heat can be observed in realized projects with a range of refrigerants, which documents the economic potential of R-718 heat pumps, implying a superior thermodynamic performance.

The relevance of designing a cost effective solution to realize the desuperheating at a minimum amount of space was outlined and different solutions were presented and compared. The liquid injection was found to be an effective solution for a precise control of the superheating, but implied significant space requirements. This might be favorable in steam supply networks, in which long piping are already in place, but it might be hindering for the integration of a heat pump into existing plants with a limited availability of space.

The suggested concept was found to be the most promising solution, offering an effective and controllable desuperheating at a minimum space. The required packing height was calculated to be less than 1 m, resulting in a small pressure drop. Comparing the required height to the recommended liquid holdup in the open intercooler of up to 1.2 m, suggests that the open intercooler might be designed more compact than suggested by Stoecker (Ch. 3.5-3.6, pp. 72-77, 1998).

Nevertheless, it is expected that the liquid holdup imposes a hydrostatic pressure being in any case larger than the pressure drop induced by the packings. The packings were chosen as a commercially available and readily applied type with ordinary characteristics and could be optimized at a later stage.

The study has focused on the evaluation of technologies to realize the intercooling between the compression stages, while the solutions might as well be applied for other working fluids or to other steam cooling applications when required. Such applications could be behind the second compression stage or in conventional steam supply systems.

5. CONCLUSIONS

The study has focused on the development of guidelines to construct two-stage R-718 heat pumps for high temperature applications. The relevance of cost and space effective solutions to realize the desuperheating was outlined and different solutions were presented. A novel approach was suggested and compared to state of the art equipment. The suggested solution consisted of a vertical vessel, which was similar to an open intercooler but filled with packings typically used in separation processes. The calculations indicated the possibility of a compact design, which can be realized with conventional and cost effective equipment. The presented concept to construct a two-stage R-718 heat pump was found to be an economically viable solution with competitive investment cost and a superior performance for applications in which the heat source is available above 100 °C.

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